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ROAD CROWN, TIRE, AND SUSPENSION EFFECTS ON VEHICLE STRAIGHT-AHEAD MOTION

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ABSTRACT – During normal operating conditions, a motor vehicle is constantly subjected to a variety of forces, which can adversely affect its straight-ahead motion performance. These forces can originate both from external sources such as wind and road and from on-board sources such as tires, suspension, and chassis configuration. One of the effects of these disturbances is the phenomenon of vehicle lateral-drift during straight-ahead motion. This paper examines the effects of road crown, tires, and suspension on vehicle straight-ahead motion. The results of experimental studies into the effects of these on-board and external disturbances are extremely sensitive to small changes in test conditions and are therefore difficult to guarantee repeatability. This study was therefore conducted by means of computer simulation using a full vehicle model. The purpose of this paper is to gain further understanding of the straight-ahead maneuver from simulation results, some aspects of which may not be obtainable from experimental study. This paper also aims to clarify some of the disputable arguments on the theories of vehicle straight-ahead motion found in the literature. Tire residual aligning torque, road crown angle, scrub radius and caster angle in suspension geometry, were selected as the study variables. The effects of these variables on straight-ahead motion were evaluated from the straight-ahead motion simulation results during a 100m run in free control mode. Examination of vehicle behavior during straight-ahead motion under a fixed control mode was also carried out in order to evaluate the validity of several disputable arguments on vehicle pull theory, found in the literature. Finally, qualitative comparisons between the simulation results and the test results were made to support the validity of the simulation results.

KEY WORDS : Tire, Road crown, Suspension, Straight ahead motion, Residual aligning torque, Vehicle drift, Vehicle pull, Computer simulation

1. INTRODUCTION

Drivers are often annoyed by vehicle behavior during straight-ahead motion due to various forces (hereinafter refereed to as *disturbances*) acting upon their vehicles. These disturbances originate from on-board as well as external sources. Some disturbances cause constant lateral deviation during straight-ahead motion. This phenomenon is called vehicle pull, steering pull or vehicle drift. The determining external causes of vehicle drift are known to be wind and road crown angle. Road crown tends to pull a vehicle toward the side of the road under straight-ahead motion. In addition to these external factors, the vehicle itself may carry its own on-board sources of disturbance due to the design specifications of its chassis, its steering system, its suspension system, and its tires. These on-board sources interact with each other. Although the literature contains several studies on the role played by tires in vehicle drift, the causal mechanisms are complex. The effect of tires on vehicle drift is characterized by the value of residual aligning torque, abbreviated to RAT (Matyja, 1987; Pottinger, 1990). However, more work needs to be done to arrive at a clearer understanding of the effects of other on-board factors such as suspension and steering systems.

Difficulties arise in studying the effects of those on-board factors experimentally due to the large number of variables involved. Moreover, the results have a strong dependency on the subjective perception of the test drivers (Stumpf, 2001). In addition, vehicle pull tendency can be expected to vary from one vehicle to another despite the fact that they have been manufactured under the same conditions. Several recent studies, undertaken to determine the manner in which tire characteristics can affect tire/vehicle system behavior in the straight-ahead maneuver, have shown that it is possible to study the effects of both on-board and external sources of disturbance on this maneuver by using computer simulation techniques (Lee, 1998; Lee, 2000).

In this paper, the effects and interactions of road

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crown, tire, and front suspension geometry on straight-ahead motion were studied using a full-vehicle model with multi-body dynamics software. A commercially available car was modeled and analyzed by the ADAMS software package (MDI, ADAMS).

This paper looks closely at the behavior of a vehicle under straight-ahead motion. Vehicle behavior was simulated using tires of different RAT values on roads of zero crown angles. Several observations found in the literature were reviewed and compared with the simulation results for the purpose of evaluating the validity of various contentious arguments.

Finally, for the purpose of qualitative validation of the vehicle drift simulation, the simulation results were compared with the road test results. For that purpose, tire RAT, road crown and scrub radius were selected as variables. Design of experiment technique was used on these variables. In this test, full factorial design was applied using two levels of value for each factor. In this case, the effect of each factor and the interaction among the chosen factors can be clearly seen.

2. VEHICLE DRIFT SIMULATION

2.1. Analysis System Model

The vehicle selected for this study was modeled as a 79 degree-of-freedom (DOF) multi-body mechanical system. Figure 1 shows a skeleton diagram of the vehicle model. This model captured the following major aspects of the vehicle; rigid vehicle body, front suspension linkage with strut and stabilizer bar, steering linkage with power booster, drive train, and rear suspension linkage with rear shock absorber.

Nine tire models were prepared with RAT values ranging from –4 Nm to +4 Nm in increments of 1 Nm. It has been proven in the literature, (Matyja, 1987; Pottinger, 1990), that tire RAT is the tire-related factor that causes vehicle pull. Tire RAT is determined by the force and moment characteristics acting on a rolling tire. Figure 2 shows three forces and three moments acting on a tire according to the SAE convention (SAE J670e). Among these forces and moments, lateral force Fy and aligning torque Mz are known to be the major influences on a tire's handling performance. Figure 3 shows lateral force and aligning torque as a function of slip angle in the range of –1 to +1 degrees. From Figure 3, RAT is defined as the aligning torque at the slip angle where the lateral force of a tire becomes zero. Therefore, the aligning torque at a slip angle α_1 represents RAT. On the other hand, the force at a slip angle α_2 is called the residual cornering force, abbreviated to RCF. Tire RAT, in Figure 3, carries a negative value. According to the SAE tire axis system, tires with positive RAT values tend to pull a vehicle to the right while negative RAT tires tend to pull a vehicle to the left.

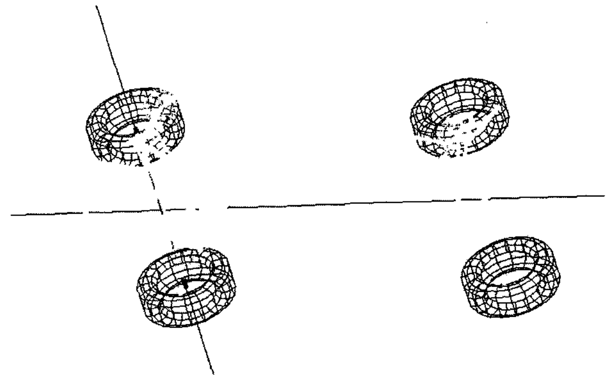

Figure 1. Skeleton diagram of the ADAMS vehicle model.

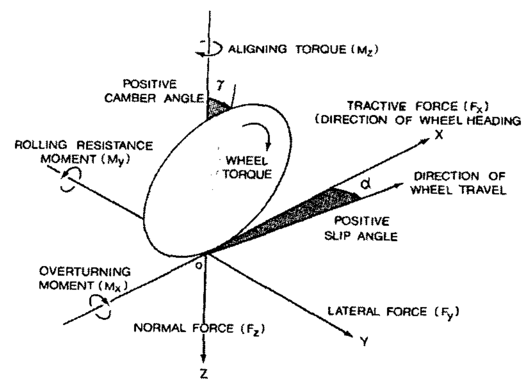

Figure 2. Forces and moments characteristics acting on a tire.

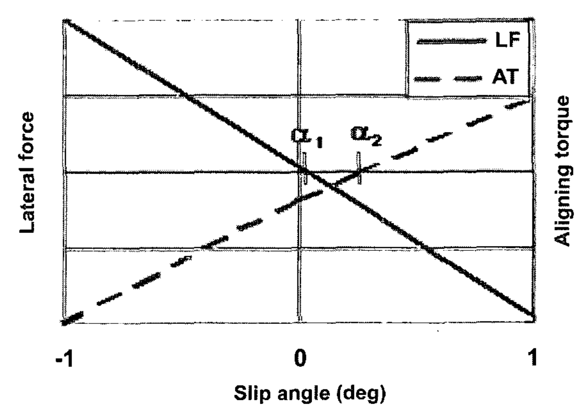

Figure 3. Lateral force and aligning moment of a tire as a function of slip angle.

Positive road crown is defined in this paper as a canted road angle with a downward slope to the right-hand side of the vehicle driver. A car pulls to the right on a road with positive road crown angle. Therefore, positive road crown is common in the countries where the left-hand drive system has been adopted. In this study, road crown angles range from –3 to +3 degrees in increments of 1.5 degrees, thus generating a total of 5 road models.

For the study of the suspension geometry effect, several different sets of suspension configurations were used. For the scrub radius effect, five sets of scrub radius configurations were designed. One way of altering the scrub radius is to change the orientation and location of the kingpin axis. However, this may cause a change in the kingpin inclination angle. The subsequent interactions between the altered scrub radius and kingpin inclination angle may affect vehicle performance. In this study, the wheel center location at the spindle was moved both inward and outward thus altering the scrub radius without causing any change in other front suspension parameters. Front left and right wheel centers were moved inward or outward within the range of –5 to +5 mm from the original design specification thus generating five sets of suspension geometries.

For the effect of caster angle change, the joints between the strut rod upper and the vehicle underbody were moved either forward by 10 mm to decrease the caster angle or backward by 10 mm to increase it. In this way, we were able to alter the caster angle without causing changes in other front suspension parameters. Due to these positional changes, the caster angle was varied by about one degree from its original design specification of 3 degrees. A total of six different suspension geometry configurations were thereby made available.

2.2. Simulation of Vehicle Straight-ahead Motion

Figure 4 shows a typical vehicle drift test track. The track consists of a control zone and a test zone. Test vehicles run through this track at a specified speed. In the control zone, the test driver adjusts the vehicle to drive in a straight-line (fixed control). The test driver then released the steering wheel in the test zone (free control). At the end of the test run, the lateral deviation from the straight line, either to the left or to the right, is measured. In order to exclude the effect of wind on the results, weather conditions should be closely monitored during a test. Simulations in this study followed the same procedure as in the vehicle drift test. During a simulation, the vehicle model was controlled to follow a pre-selected straight line at a constant speed of 80 km/h. The vehicle model ran through for a distance of 100 m in a fixed control mode. The vehicle then ran a further 100 m in a free control mode with the steering wheel released. Lateral drift during the 100 m straight-ahead run in free control mode was recorded.

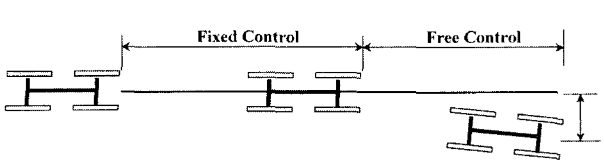

Figure 4. Vehicle drift test during a straight-ahead motion.

3. SIMULATION RESULTS OF VEHICLE STRAIGHT-AHEAD MOTION

3.1. Tire RAT and Road Crown Effect

For the study of the tire RAT effect on vehicle drift, nine tire models were fitted to the vehicle model. The nine-tire/vehicle system models ran consecutively through on five different road surfaces. Figure 5 shows the simulation results. Of the 45 simulations conducted, two failed due to convergence problems and were not included in the overall results. As previously mentioned, these results measured the extent of lateral drift after a 100 m run under free control mode. In the simulation results, a positive value represented a drift to the right. As can be seen from this figure, the lateral drift of a vehicle is directly proportional to the values of tire RAT and the road crown angle.

Figure 5 also shows that there was no interaction between tire RAT and road crown. From these results, we can calculate a proper range of RAT values for a given road crown angle. If, for example, we choose a road

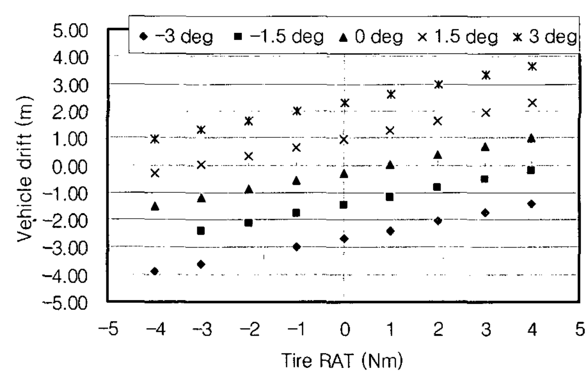

Figure 5. Simulation results of vehicle drift due to tire RAT and road crown.

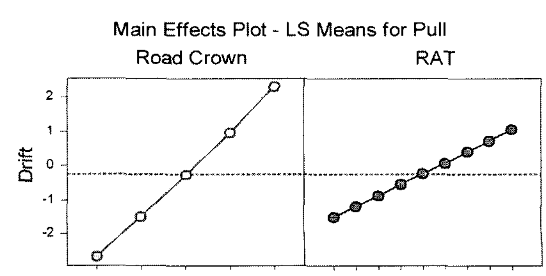

Figure 6. Effect of road crown and tire RAT on the vehicle drift simulation.

crown angle of zero degrees, and an acceptable lateral drift of −1.5 m to +1.5 m, a proper RAT range would be within the range of −4 Nm to +6 Nm. Vehicle drift sensitivity to tire RAT was calculated to be 0.314 m per 1 Nm RAT change, while drift sensitivity to road crown angle was found to be 0.828 m per 1 degree of road crown angle change. Figure 6 shows the net effect of road crown and tire RAT on vehicle drift.

3.2. Validation of Vehicle Pull Theories in the Literature

Toppings (Toppings, 1975), Dijks (Dijks, 1981), and Lee (Lee, 2000) have shown that vehicle attitude angle during straight-ahead motion can be affected by tire characteristics. Lee (Lee, 2000) calculated parameters related to vehicle attitude during straight-ahead motion under fixed control mode using a two-wheeled vehicle model shown in Figure 7. These parameters were calculated as follows:

$$\alpha_f = \frac{-(a+b)CF_{yo} - 2KF_{yo} - 2CM_{zo}}{-(a+b)C^2} \tag{1}$$

$$= \alpha_1 + 2\frac{K}{(a+b)C}(\alpha_1 - \alpha_2)$$

$$\alpha_r = \alpha_1 - 2\frac{K}{(a+b)C}(\alpha_1 - \alpha_2) \tag{2}$$

In these equations, angles α_f and α_r represent the slip angles at the front and rear axles, slip angles α_1 and α_2 are as defined in Figure 3, C and K represent cornering stiffness and aligning torque stiffness, and $(a+b)$ represents the vehicle's wheel-base. F_{y0} and M_{z0} are the lateral force and aligning torque at zero slip angle, respectively. Since there is no yaw velocity during straight motion under the fixed control mode, the sideslip angle ψ of a vehicle is the same as the slip angle α_r at the rear axle.

Table 1 compares the results from a calculation with

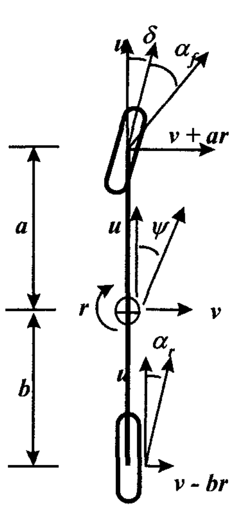

Figure 7. Two-wheeled vehicle model and its motion variables.

Table 1. Simulation and analysis results for vehicle attitude during straight-ahead motion under fixed control mode. Unit : degree

Tire RAT		−4Nm	−3Nm	−2Nm	−1Nm	0Nm
Slip angle at the front zxle	Sim.	0.192	0.192	0.193	0.193	0.193
	Calc.	0.209	0.210	0.210	0.211	0.211
Slip angle at the rear axle	Sim.	0.191	0.190	0.190	0.189	0.189
	Calc.	0.213	0.213	0.212	0.212	0.211
Vehicle sideslip angle	Sim.	0.190	0.190	0.190	0.189	0.189
	Calc.	0.213	0.213	0.212	0.212	0.211

the two-wheeled vehicle model and a simulation with the full vehicle model. In equations (1) and (2) above, the value of cornering stiffness times wheelbase is much bigger than that of aligning stiffness. Therefore, we can expect that the slip angle at the front axle, the slip angle at the rear axle, and the vehicle sideslip angle are mainly dependent upon the slip angle α_1 at which the lateral force at the front axle becomes zero. The tire model in the simulation has an α_1 value of 0.211 degrees. From Table 1, we noticed that this angle is the main factor controlling the slip angle at the front axle, the slip angle at the rear axle, and the vehicle sideslip angle during a straight-ahead motion under the fixed control mode. There are some discrepancies between the computer simulation results and the mathematical analysis of the two-wheeled vehicle model in Table 1. These are thought to derive from the suspension compliance effect on the vehicle model.

For the analysis of straight-ahead motion under the fixed control mode, some references (Pottinger, 1990; Toppings, 1975) had assumed that, during straight-ahead motion under the fixed control mode, there was no lateral force acting on the front axle. In a reference by Lee (Lee, 2000), from the analysis of dynamic equation of two-wheeled vehicles, the summation of lateral forces acting on front and rear axles was put at zero. From the simulation results of this study, it was found that the summation of lateral forces acting on the front and rear axles was approximately zero as shown in Table 2. For the free control mode, some references have assumed that

Table 2. Lateral forces acting on each spindle during straight-ahead motion under fixed control mode. Unit : N

Tire RAT	−4 Nm	−3 Nm	−2 Nm	−1 Nm	0 Nm
Front left	−36.9	−37.5	−38.0	−38.6	−39.2
Front right	43.0	42.4	41.8	41.2	40.6
Rear left	459.7	460.3	460.8	461.4	462.0
Rear right	−465.9	−465.3	−464.9	−464.2	−463.6

the aligning torque at front axle was zero. The same results were obtained from the simulation in this study. That is to say that, as expected, the aligning torque at the steering wheel was zero during straight-ahead motion under the free control mode.

3.3. Scrub Radius and Caster Angle Effect

In Table 3, the results for the five configurations used in the study of the scrub radius effect are summarized. Figure 8 shows the simulation results with these configurations of scrub radii. Again, a negative value in the simulation results meant a drift to the left, and a positive value indicated a drift to the right. As can be seen from the simulation results, if symmetry of the scrub radius between the left and the right wheels at the front axle is preserved, there is no significant change in vehicle drift. However, if we fail to preserve symmetry in the scrub radii, the vehicle drifts laterally in the direction of the wheel with the less positive scrub radius. This phenomenon can be explained by considering the geometry of the suspension linkage. As the joint location between the wheel center and the spindle moves outward, the moment, due to the axle load around kingpin axis, changes. If symmetry in scrub radii between the left and the right wheel suspension is maintained, the moment between the left and right wheel is balanced. Therefore, there will be no change in vehicle drift. If symmetry is violated, the vertical load at the wheel develops a moment to rotate the steering wheel in the direction of the wheel with a less positive (or more negative) scrub radius. Due to the moment imbalance around the kingpin axis, the vehicle starts to drift in the direction of the wheel with a more negative scrub radius.

In addition, we observed that the sensitivity of the vehicle drift to tire RAT remained unchanged throughout all of the configurations of scrub radii in Table 3. Therefore, we can conclude that there is no interaction between scrub radius and tire RAT. Refer to Figure 8.

Table 4 shows the six configurations of the caster design used in this study. With these configurations of strut, six simulations were performed on the flat road surface. Figure 9 summarizes the simulation results. The simulation results showed that the more positive the caster angle of the front axle was, the less drift the vehicle had. However, if the symmetry between the caster angles of front left and right wheels is preserved, the caster angle had no significant effect on a vehicle's drift tendency. In view of this, simulation results with strut 1 and strut 2 configurations were not shown in Figure 9. Instead, "No

Table 3. Scrub radius configuration for each simulation.

	Scrub radius configuration
Scrub1	Front left and right wheel moved 5 mm outward
Scrub2	Front left wheel moved 5 mm inward, front right wheel moved 5 mm outward
Scrub3	Front left wheel moved 5 mm outward, front right wheel moved 5 mm inward
Scrub4	Front right wheel moved 5 mm outward
Scrub5	Front left wheel moved 5 mm outward

Table 4. Strut configuration for the study of caster angle effect.

	Strut configuration
Strut1	Front left and right joint move forward By 10 mm (Caster Angle : 2.18 deg)
Strut2	Front left and right joint move forward by 10 mm (Caster Angle : 3.75 deg)
Strut3	Front left joint moved forward by 10 mm
Strut4	Front left joint moved backward by 10 mm
Strut5	Front left joint moved forward and right joint moved backward by 10 mm
Strut6	Front left joint moved backward and right joint move forward by 10 mm

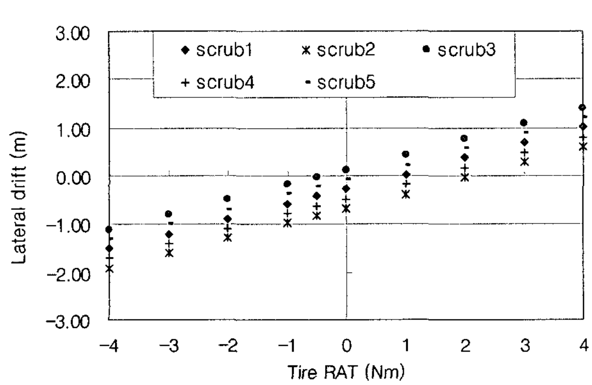

Figure 8. Effect of Scrub Radius in Vehicle Drift Simulation.

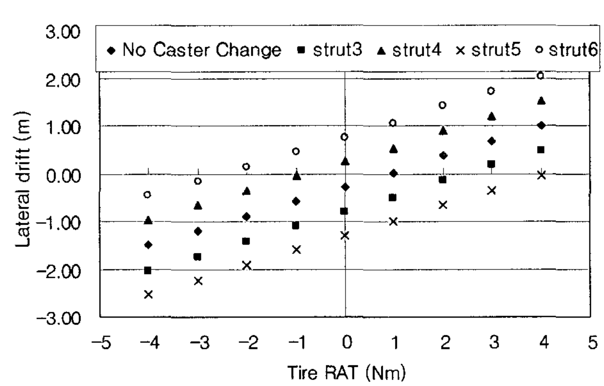

Figure 9. Effect of Caster Angle in Vehicle Drift Simulation.

Caster Change" is shown since this result is almost the same as the results with strut 1 and strut 2 configurations. If, however, the symmetry between left and right front wheels is violated, the vehicle drifts laterally in the direction of the wheel with less positive (or more negative) caster angle. It can be seen from the simulation results that there was no interaction between the caster angle and tire RAT.

4. VALIDATION OF SIMULATION RESULTS

4.1. Test and Simulation Program

For the validation of the simulation results of the previous section, another set of tests and simulations was scheduled. In this validation process, road crown, scrub radius and tire RAT were selected as study variables in the design of experiment (DOE). Each variable had two levels of value. From the available techniques in DOE, full factorial design was applied since it allows us to clearly see the effect of each variable and the interaction among the design variables.

Table 5 shows the full factorial design for the vehicle drift test. Tires with RAT values of –0.5 Nm and –3 Nm were selected. For the change of scrub radius, a spacer, with a thickness of 5 mm, was inserted between the axle hub and the wheel. If the spacer was inserted at the front left axle, then the scrub radius difference (or cross scrub radius) was +5 mm. At the front right axle, cross scrub radius was –5 mm. For the road crown angle, the same sign convention used in section 2.1 was applied.

For the simulation, instead of having new models for tire RAT values and road crown angles according to the test program, the simulation model outlined in section 2.1 was used. Although we did not use the models with the same tire RAT and road crown values as those in the vehicle test, we inferred the simulation results for tire RAT and road crown level. This was due to the full factorial design on the simulation variables. The full factorial design for vehicle drift simulation is shown in Table 6. Test results and simulation results based on full factorial design were analyzed by means of MINITAB (Minitab Inc., 2000) statistical software.

Table 5. Full factorial design of variables and test results.

Set	Road crown (deg)	Scrub radius (mm)	Tire RAT (Nm)	Drift (Cm)
# 1	–0.5	–5	–3.0	–152.0
# 2	+0.5	–5	–3.0	–46.0
# 3	–0.5	+5	–3.0	–73.0
# 4	+0.5	+5	–3.0	–8.3
# 5	–0.5	–5	–0.5	–107.5
# 6	+0.5	–5	–0.5	–17.5
# 7	–0.5	+5	–0.5	–42.2
# 8	+0.5	+5	–0.5	32.3

Table 6. Full factorial design of variables and simulation results.

Set	Road crown (deg)	Scrub radius (mm)	Tire RAT (Nm)	Drift (Cm)
# 1	–1	–10	–3.0	–241.7
# 2	+1	–10	–3.0	–80.0
# 3	–1	+10	–3.0	–161.4
# 4	+1	+10	–3.0	14.0
# 5	–1	–10	0	–149.2
# 6	+1	–10	0	13.5
# 7	–1	+10	0	–68.2
# 8	+1	+10	0	95.9

4.2. Test and Simulation Results

Figure 10 shows the effects of each test variable, while Figure 11 shows the interactions among the design variables from the vehicle drift test results. As can be seen from Figure 10, road crown has the most significant effect on vehicle drift within a selected range of test variables. Although there may be a slight interaction between scrub radius and road crown, we can conclude that there is negligible interaction between tire RAT and scrub radius, between tire RAT and road crown and between scrub radius and road crown. The estimated regression equation of the effect of these variables on vehicle drift test was derived from MINITAB as follows:

$$\begin{aligned} \textit{Drift}\ (\text{cm}) = &-26.5 + 81.6 \times RC + 2.87 \times SR \\ &+14.45 \times RAT - 0.517 \times RC \times SR \\ &-1.234 \times RC \times RAT \\ &-0.015 \times SR \times RAT \\ &+0.517 \times RC \times SR \times RC \end{aligned} \tag{3}$$

where RC represents road crown and SR represents scrub radius.

Figure 12 shows the effects of each simulation variable and Figure 13 shows the interactions among these

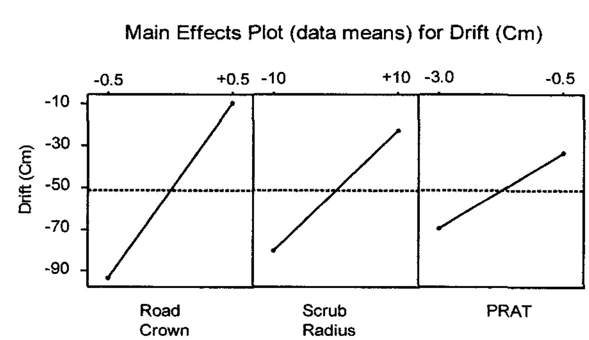

Figure 10. Effect of design variables used in the vehicle drift test.

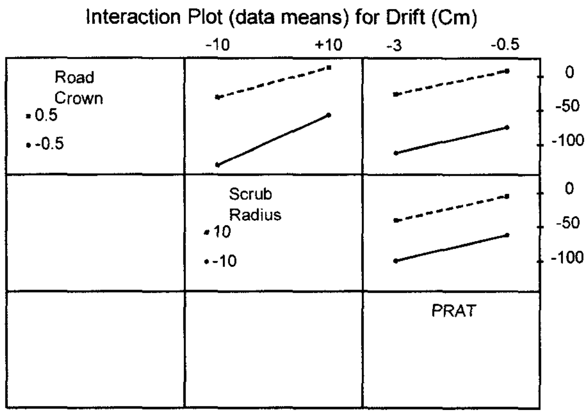

Figure 11. Interactions among the design variables in the vehicle drift test.

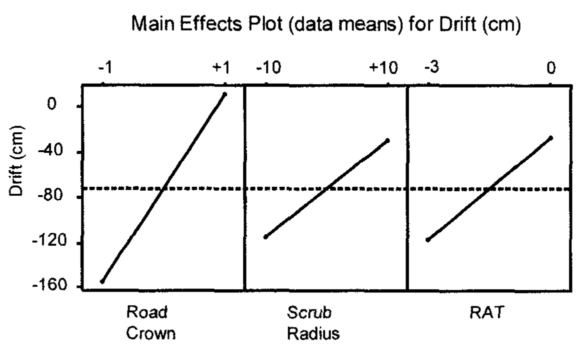

Figure 12. Effect of design variables used in the vehicle drift simulation.

variables from the vehicle drift simulation results. As can be seen from Figure 12, road crown has the most significant effect on vehicle drift within a selected range of variables. In Figure 12, we can conclude that there is no interaction between tire RAT and scrub radius, between tire RAT and road crown and between scrub radius and road crown. This confirms the simulation

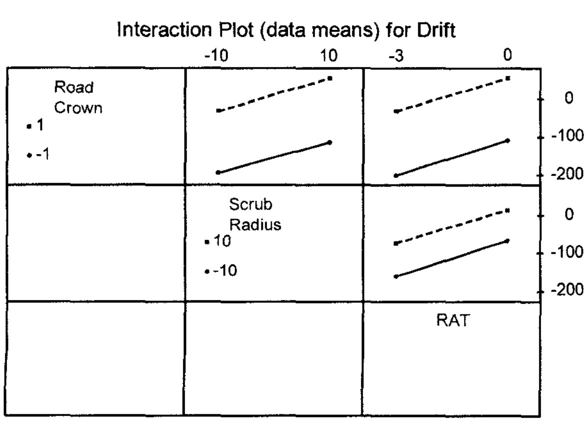

Figure 13. Interactions among the design variables in the vehicle drift simulation.

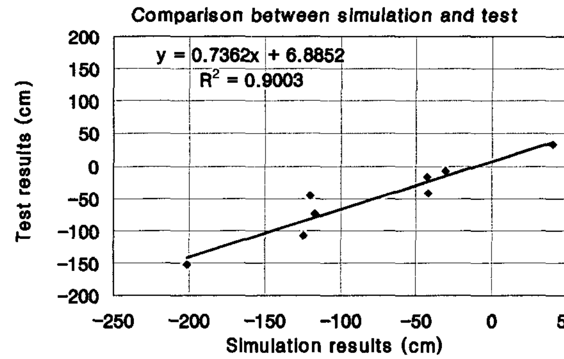

Figure 14. Comparison of vehicle drift between simulation and test.

results in the previous sections. The estimated regression equation of the effect of these variables on vehicle drift test results was found from MINITAB as follows:

$$\begin{aligned} Drift\ (\text{cm}) = & -27.0 + 81.7 \times RC + 4.09 \times SR \\ & +30.1 \times RAT - 0.04 \times RC \times SR \\ & -0.86 \times RC \times RAT \\ & -0.091 \times SR \times RAT \\ & +0.103 \times RC \times SR \times RC \end{aligned} \quad (4)$$

Using equation (4), we were able to calculate the simulated amount of vehicle drift for the cases of tire RAT values and scrub radius values used in the tests. In Figure 14, the simulation and test results for vehicle drift are compared. The simulation results in this comparison were calculated by plugging the same tire RAT and scrub radius values used in the tests into equation (4). Although Figure 14 shows a good correlation between simulation and test outcomes, the absolute drift values from the simulation results overestimate the drift values from the tests. This overestimation is partially due to the bushing and spring effect in the suspension and steering system. The test vehicle used was almost three years old and had a travel record of approximately 40,000 km. Therefore, the stiffness of the bushing rubber in the suspension and steering linkage is relatively low compared to that of a new vehicle. Similarly, stiffness of the spring on the strut is expected to be relatively low compared to that of a new vehicle. Based on the authors' experience, changing the spring constant and the bushing rubber stiffness in a vehicle model to lower values leads to reduced vehicle drift in simulation.

5. CONCLUSIONS

In this paper, the effects of tire, road crown, and vehicle suspension on vehicle straight-ahead motion were studied using computer simulation. For the study of tire effect, residual aligning torque was selected as a study variable. Simulations were performed on a vehicle model with nine different tire models ranging in RAT from –4 Nm to +4 Nm. Road crown angle was varied from –3 degrees to +3 degrees. Scrub radius and caster angle were chosen for the study of suspension effect. For the scrub radius effect, five sets of scrub radii were designed so that scrub radius varied in the range of –5 mm to +5 mm with respect to the original design. For the caster angle effect, six sets of suspension configurations were designed by moving the joints between the strut rod upper and the vehicle underbody. These joints moved forward or backward by 10 mm to induce about 1 degree change in caster angle.

From the forty-five simulations carried out with a full combination of nine tire RAT models and five road crown angles, it was found that the vehicle drift was directly proportional to tire RAT and road crown. There was no interaction between tire RAT and road crown. In this study, the vehicle has a sensitivity of 0.314 m per 1 Nm tire RAT change in the front axle and 0.828 m per 1-degree road crown angle change. In the fixed control mode, the vehicle exhibited sideslip angle. This sideslip angle is dependent mainly upon the slip angle of the front axle tires where lateral force becomes zero. By comparing simulation results with the theories in the literature, it was found that the total lateral forces acting on the front and rear axles become zero when a vehicle is running straight-ahead under fixed control mode.

From the study of the suspension effect, it was shown that there was no significant change in the vehicle behavior during a straight-ahead motion unless suspension geometry failed to keep symmetry between the left and the right front wheels. If the scrub radius of one wheel was more positive in its value than that of the other, then the vehicle drifted in the direction of the wheel which had a less positive scrub radius. The amount of vehicle drift due to scrub radius was directly proportional to the difference between the scrub radii of the left and right wheels at the front axle. A similar result was obtained in the simulation of caster angle change. If the caster angle of one wheel was greater than that of the other, the vehicle drifted laterally in the direction of the wheel which had a smaller (or more negative) caster angle. The amount of vehicle drift was directly proportional to the difference between the caster angles of the left and the right wheels.

In order to achieve a qualitative presentation of the validity of the simulation results, comparisons between simulations and tests were made on separate sets of test and simulation results. In this program, road crown, tire RAT, and scrub radius were selected and full factorial design in DOE was applied for the study of variable effect in test and simulation. The results show that, although the simulation results overestimate the test results, both sets of results showed the same tendency in vehicle drift due to variable changes.

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